PRESSURE BALANCED, RADIALLY COMPLIANT NON CONTACT, SHAFT RIDING SEAL

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John Short discussed some of the work EG&G R&D is doing on brush seals. We have a parallel program which I call the pressure balanced radially compliant non-contact shaft riding seal. And non-contact is the big thing we are trying to achieve here. I will discuss what we have accomplished in the last 12 month period We have managed to pressure balance this seal so that now we can get hydrostatic lift off which is not speed dependent. To put numbers on that, we can put 70 psi on the seal and turn it with two fingers with absolutely zero torque and zero contact. We have a mechanism that gives us a moment balance that will optimize the finger lay down on the shaft, I'll demonstrate that. We are trying to achieve film stiffness improvement. We are analyzing the structure and looking at its stiffness. I might mention that for this sort of device, for it to work with non-contact over a wide range of closures, we want to maximize the film stiffness and probably the structural stiffness. It turns out that the ratio of the film stiffness over your structural stiffness is the same as the ratio of the delta R you can take for a given change in thickness of your operating film.

And I will talk about a triple ply seal that we built and tested. Where we are right now with this technology is we have gone from 0 to 60 psi in 5 mils delta R of closure, 12,000 rpm. This was with absolute non-contact as measured electrically. We also achieved 0 to 60 psi with 9 mils delta R, 1200 rpm with very light contact, about .25 inch-lb of torque. These are nominally 4 3/4 diameter seals. What do these look like? What the pressure balanced compliant seal is, you can see that it is a plurality of thin metallic sheets of metal. Pressure is on this side of the seal and we have non-restrictive orifices so this cavity contains full system pressure. Basically these are the shaft riding fingers. We also have two additional layers of fingers which are over lapped, such that all the slots are blocked. So the only slot flow you have is the through the 8 mil path curving down through the shaft riding fingers and you are totally blocking the radial flow through the slots. You have the sealing dam on this side. What this might look like in a cross section, you can see there, would be like this. Again, just to put numbers on it, this was a 4 3/4" diameter shaft, by 5/8" axial length seal, we were putting systems pressure on this side of about 70 psi. These are two rows, this is your seal dam and this is the shaft riding fingers lying on a film between its bore diameter and the shaft. Now this is a hydrostatic device.

That means, that when you just put pressure on, even before you put on speed, it will deform to maintain an operating film "H". Our goal of course, as time goes on is get H down to very small levels, like maybe 2 to 3 tenths of a mil. Right now my computer codes are telling me we are running at about 7 to 8 tenths of a mil. I hope to show you some of that. So basically this is a feed back device, even non rotating. The shape of the pressure drop at the top of the fingers is always constant, it just goes up into the secondary seal, no more or less. Then there is very little pressure drop in what I call the bridge section, since clearances are very large; typically .030 inches. Most of the pressure drop is over the Flat1 section from the heal to the tip. At the design clearance pressure would drop along some parabolic path on the non-rotating position. Now if any mechanism such as shaft excursions or whatever try to make the clearance smaller, it would result in the solid line being shifted as shown by this dotted line, creating more opening for us just like non contacting gas face seals. Vice versa, if your clearance would try to increase then the larger H would have a corresponding pressure distribution like this, therefore the closing force would be greater and you could go back down to your design clearance. This only gets better when you run the seal at speed, because hydrodynamic forces will also be obtained.

We have investigated about 3 different hydrodynamic lift off mechanisms. The first one I'll talk about, is mismatched curvature and I am exaggerating this drawing. As you can see, with this design the seal bore has a slightly larger radius than the radius of the shaft. If it were to come down and touch right here it wouldn't quite fit at the edges and you would get a little bit of mismatched curvature. Of course, the seal would still be concentric but the minute you rotate the shaft, what would happen is you would be pulling gas into this converging wedge causing an increase in pressure and sucking gas out of the diverging wedge. Which would actually cause a decrease in pressure that would cause the center of the pressure to shift over and cause a tipping action of the entire finger. So we obtain a larger entrance clearance and a smaller exit clearance. So that is one mechanism, geometric mismatch curvature depending on the width of the seal and the differential radii isn't a lot. It might vary between maybe 20 micro inches and 150 micro inches depending on the diameters and radii and so forth.

The other good mechanism we have here is thermal deflection. What happens here is we heat the underside of the finger due to the shear going on, dissipating energy and you get a temperature drop through the thickness of the finger which also gives you the thermal deflection which increases your mismatch curvature. ??? All of which tends to cause the seal to tip and form a converging edge in the direction of rotation. The third mechanism is that of angle cut fingers. We also are building these types of seals with angle cut fingers. The thing kind of wraps around the shaft so we're going to get a foil bearing type of lift off. I will be investigating that experimentally over the next several months. As I mentioned, it is very important to get good film stiffness so that these fingers will lay down on the shaft as flat as possible or maybe with just a little bit of a converging wedge in the direction of pressure. It turns out that this little tip extension gives us an angular mechanism, but tends to give us a parallel layout condition. I demonstrate this with this kind of an arm waving type of figure here. It is not quantitative it is only qualitative.

But basically you can see nothing is happening in the bridge section because it is the same pressure here and here. If you take the center of pressure, of the closing force, it never changes. But if you take the center of pressure of the opening force, if it were here for a parallel position if you then had an excursion it would tend to make the tip lie down like this, nothing happens on the top of the seal. If the center of pressure stays put, but if the seal angles down like this, the old center of pressure is going to be moved forward causing a moment M3 which tends to partially give you a restored lay down position. Not perfect, but that is the mechanism and conversely if your shaft were to try to expand relative to the seal, either centrifugally or thermally and you would tend to want to hit the heal of the Flat1. You could easily see that this would drastically pinch p off the pressure drop under the seal causing a very large shift of the center of pressure, and giving a moment in the other direction which again will cause a partial restored lay down. What we are doing now is trying to analytically and iteratively compute the shape and lay down of the figure relative to the tip direction and circumferential direction. We have spent quite some time trying to analyze these seals. We developed an internal code, COMP 3, that is represented here. We can input the finger geometry, various clearances, various thermal and/or geometric mismatch and then by applying shaft rotation we can look at the pressure distribution. What you are looking at here is the axial direction so you are going from about 60 psi down to 14.7 psi at the exit. With the direction of shaft rotation this way, you are getting a hydrodynamic pumping up. This code, however, is not iterative so we can essentially guess at the clearances, guess at the lay down and you now get film stiffness and you know the pressures. But what we are doing now, which I will describe, is going towards an iterative solution. Before I get into that I'll talk about me last few slides to try and not take up too much time.

The Test Rig is kind of a poor drawing; it is a 20,000 rpm motorized spindle, I won't leave this up. We have hydraulic shifters that are similar to the ones that John Short was showing for varying the delta R. It is easier for us to move the seal case and leave the shaft rotating to simulate closure. We measure the shaft vibration with Bentley probes, we have thermocouples, we measure mass flow and then we shake the seal case relative to the shaft with hydraulic shakers similar to what John was showing. Just for a quick leakage comparison of where this technology is today, I compare it with a straight toothed laby seal having ten teeth, it is the typical industrial one. It would have a flow parameter which would go to about 5 ½, actually .0055. You will find that a flow parameter is mass flow in pounds per second times the square root of T in Rankine divided by upstream pressure, psia and diameter in inches. It turns out, of course, that a laby seal with a 6 mil radial clearance has about a flow parameter of about .006. The second curve is a 4 tooth aerospace laby seal. This is experimental data. For a 30,000 rpm seal about 600°F, operating clearance of 6 mils, there is a bit lower leakage. Figure 3 would be a 1995 version brush seal, we are a little better than that now. But basically this would be the flow parameter of the EG&G experimental brush seal at 900 ft/s and 428°F. And this forth curve is the experimental test leakage of the compliant seal that I described where you can see it takes about 5 psi to get the fingers to come down, but then we achieve a flow parameter of about 1 ½ which of course is .0015 in the actual equation.

Lastly, I have 2 slides. We now have an operational iterative solution. This is a Reynolds number solution and understand for air we have to be on films of less than about 4/10 of a mil to be accurate. We are, however, looking at ways of putting inertia into the element. What we did here, we actually developed an element which is combination of a fluid and a structural iterative element that we are inputting into ANSYS so we have the power of ANSYS for modeling any structure, shape, angle or anything. Of course, we will be putting in an additional capability modeling not only the finger that lies on the shaft but the covers, the secondary seals and all its detail. This is relatively easy to do compared to what has been done before. This is the seal we built and tested. This seal was run at 70 psi, non-contact up to 5 mils delta R. This slide shows that this particular seal was running at fairly thick clearances, about 2 mils in this region and about 8/10 mil in this region. But the analysis confirms that the seal did run at 70 psi. These are the pressure contours. You can see that these are almost the hydrostatic solutions. Not much hydrodynamics going on here. There is no pressure drop in the bridge section and you start to get this pressure drop in the flat1 region. And corresponding to the flat1 pressure drop is a pressure contour. This is actually a deflection map. I do not have it with me but as of today we have film stiffness maps as well, but basically these are the deflections.

The seal started out centered on the shaft with a 7 ½ mil radial clearance. You can see that the deflection of the tip which is about 6 mils would leave a clearance of about 1 ½ mils. And of course at the place where it is welded to the housing there is no deflection, that means you still have your original clearance. So we finally do have an iterative code running. We have other work to do, we want to put in the thermal deflection and make it iterative. Right now I can calculate thermal deflection but I have ro use it as an outside input to the code. We will make it iterative as well. We will put ion the details of the slot cover. At the secondary seal we use the same element between the secondary seal and the seal dam. That is to make sure we do not lock up in that region. I think it will be a very powerful design tool. We are working on the seal both for air applications, air to oil applications and solid oil applications.

QUESTIONS

Q. Can the scale of the seal out to bigger diameters and larger deflections?

A. Yes, we feel we are going to be able to do that. We actually have a few development contracts for the seal going up to about 20 plus diameter inches. I think that it is essential we get the test results on our aerospace rig up to about 1100ft/sec. Which is about what you are looking at when we make the leap up to the twenty, twenty-four inch diameter applications.

COMPLIANT SEAL
95 TECHNICAL MILESTONES

I. **HYDROSTATIC LIFTOFF**
(NOT SPEED DEPENDENT)

II. **MOMENT BALANCE**
(OPTIMIZE FINGER LAYDOWN ON SHAFT)

III. **FILM STIFFNESS** (IMPROVEMENT)

IV. **STRUCTURE** — DIFFERENTIAL STIFFNESS

V. **NEW TRIPLE PLY**
TESTED
(@ 0—60 PSIG, .005 ΔR,12000RPM, NON CONTACT)

(@ 8—60 PSIG, .009 ΔR,12000RPM, LIGHT CONTACT)

VI. **VERSION IV**
(TO BE TESTED THIS MONTH)

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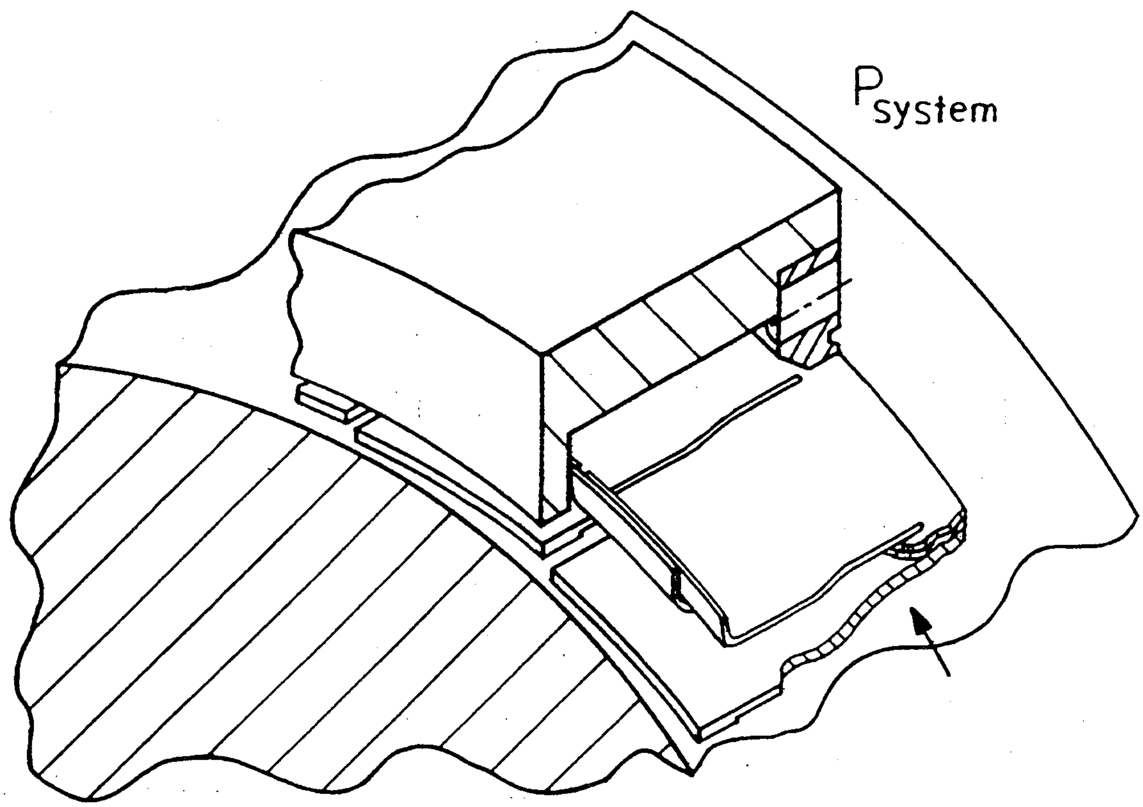

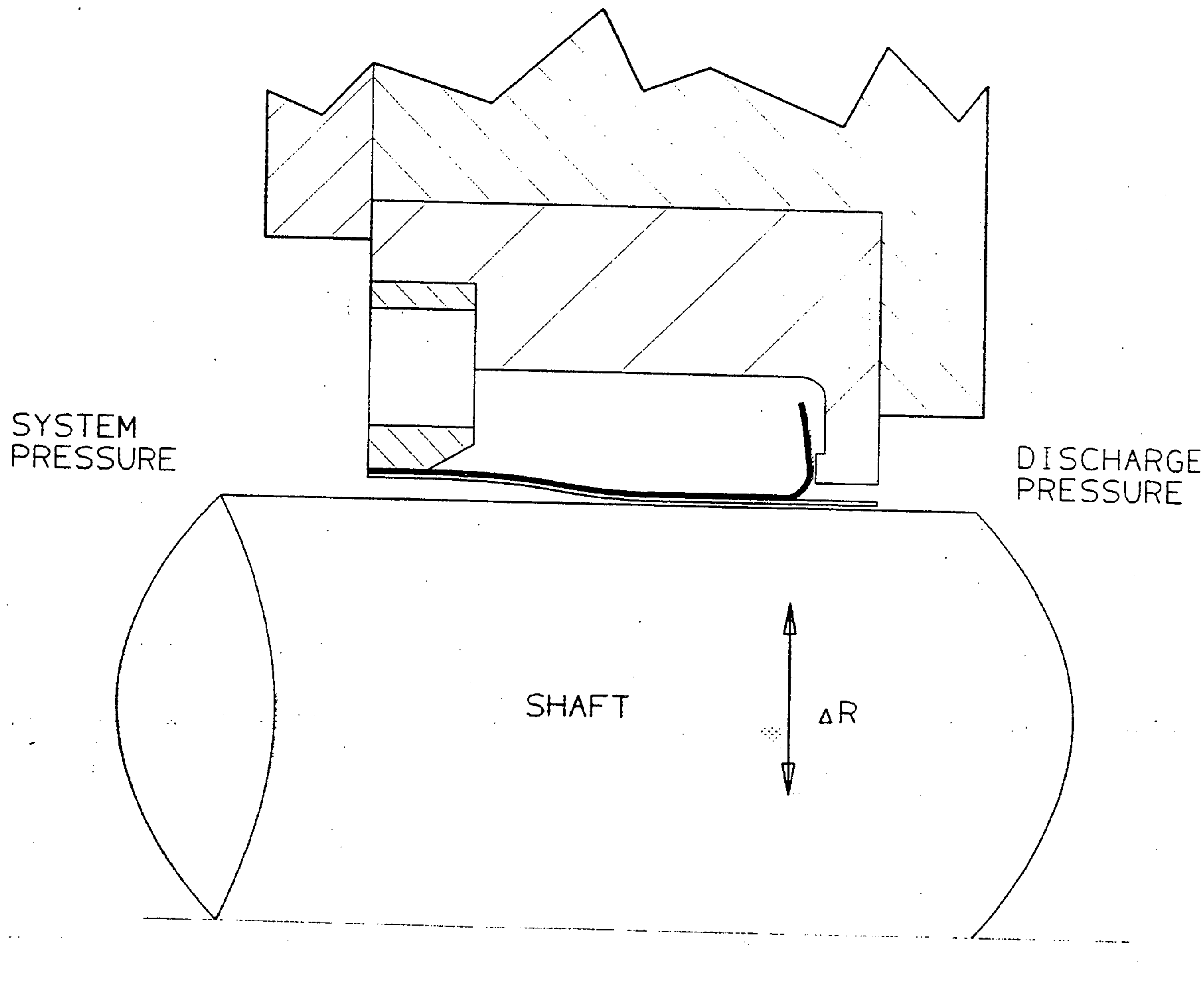
SYSTEM
PRESSURE
DISCHARGE
PRESSURE
SHAFT
ΔR

EG&G

Description: (Please read instructions on reverse side)

TYPICAL PRESSURE DISTRIBUTIONS
(NON-ROTATING)

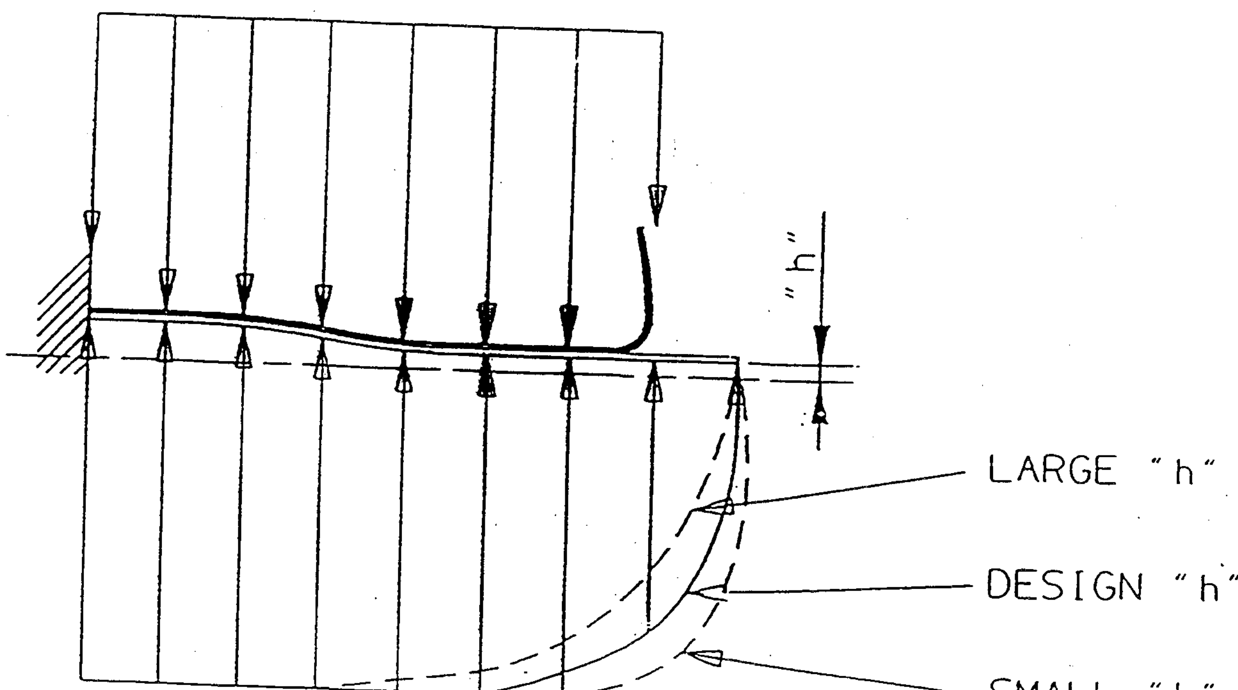

FIGURE 2

EG&G

;cription: (Please read instructions on reverse side)

TYPICAL PRESSURE DISTRIBUTIONS
(ROTATING CONDITION)

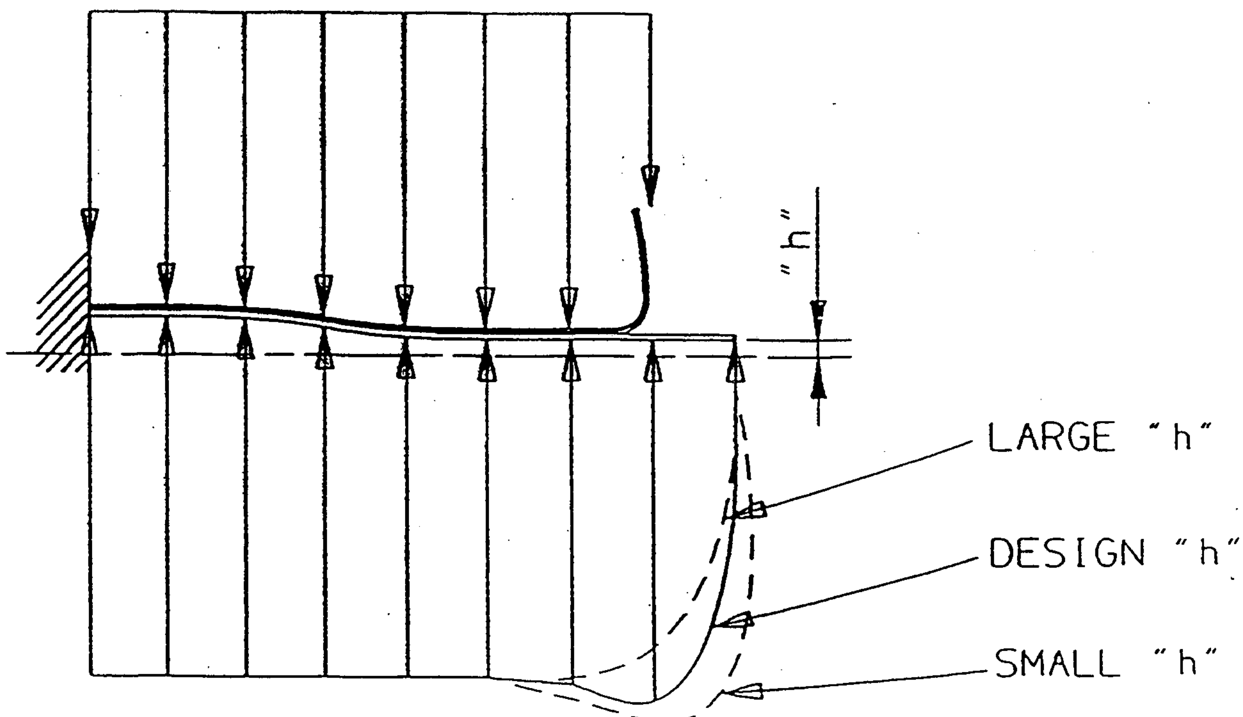

FIGURE 3

Typical Pressure Distribution for Straight Cut Compliant Seal without Slot Cover

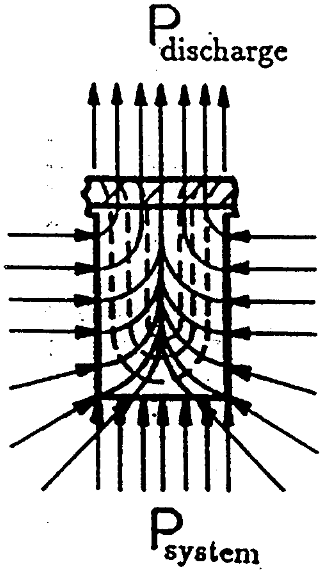

Figure 2a

dotted lines represent lines of constant pressure between the shaft the underside of the compliant metallic members

solid lines represent the direction of the flow field

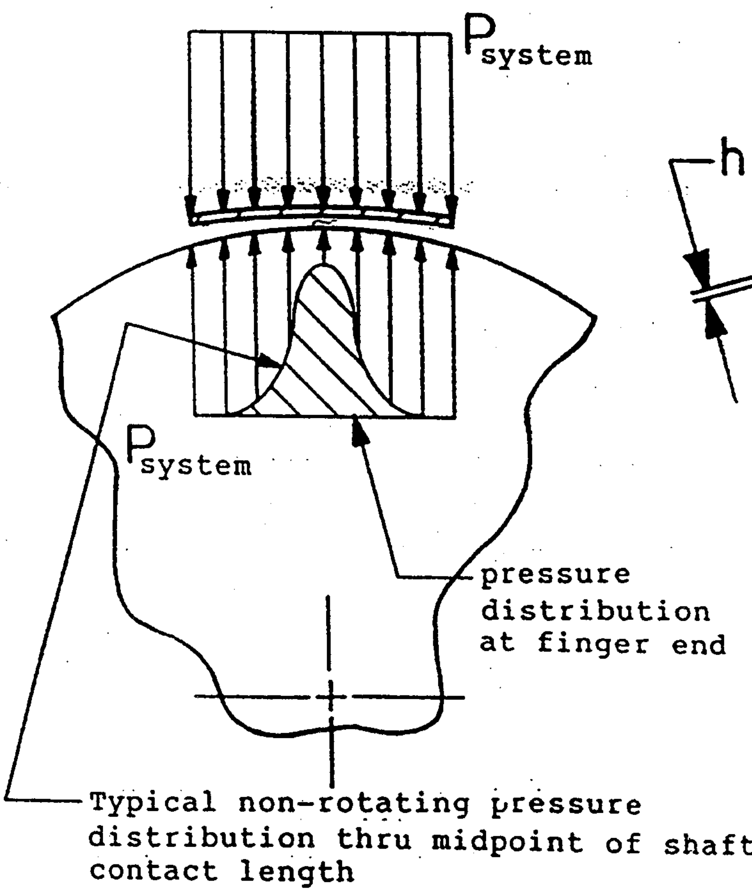

Figure 2b

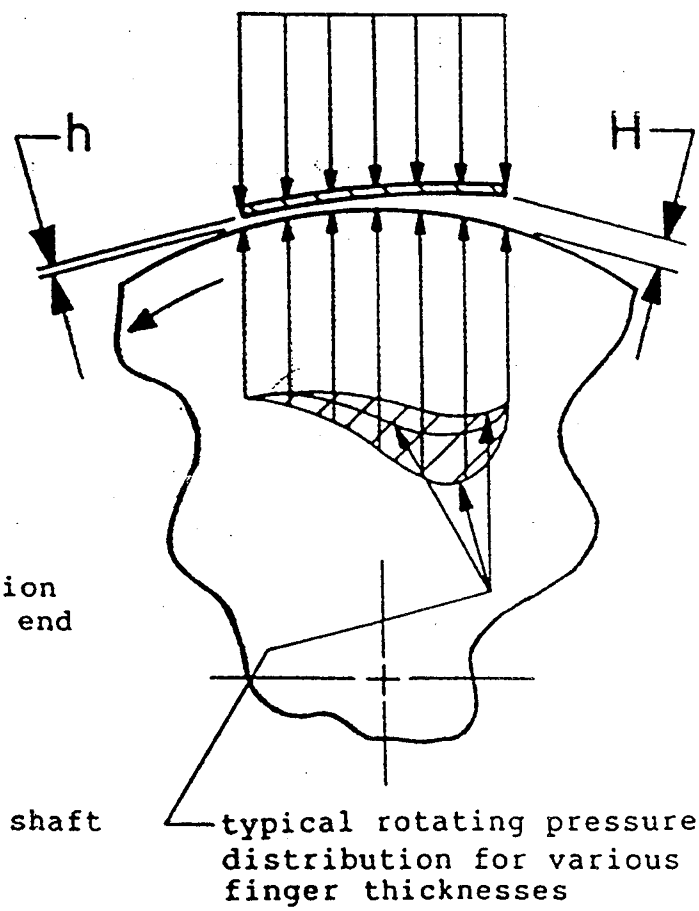

Figure 2c

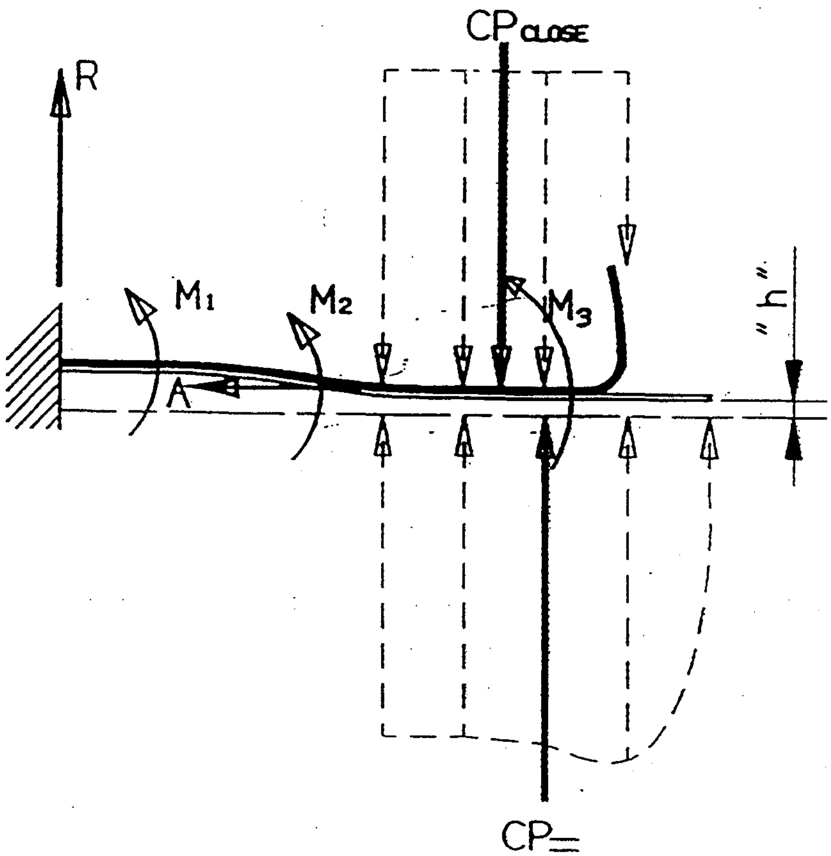

FIGURE 5: PARALLEL CONDITION. BALANCE STATE

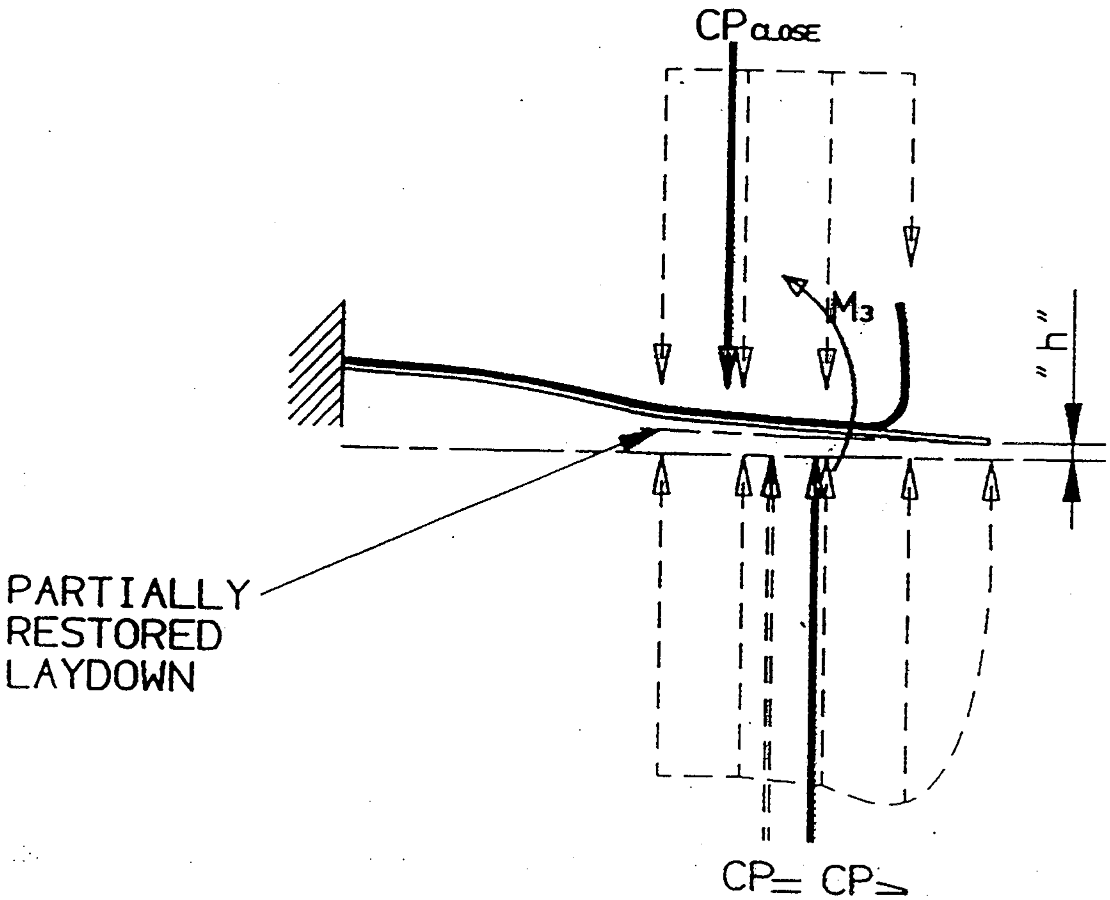

FIGURE 6: CONVERGING WEDGE CREATES MOMENT "M_3". WHICH WIL
A PARTIAL RESTORATION OF LAYDOWN.

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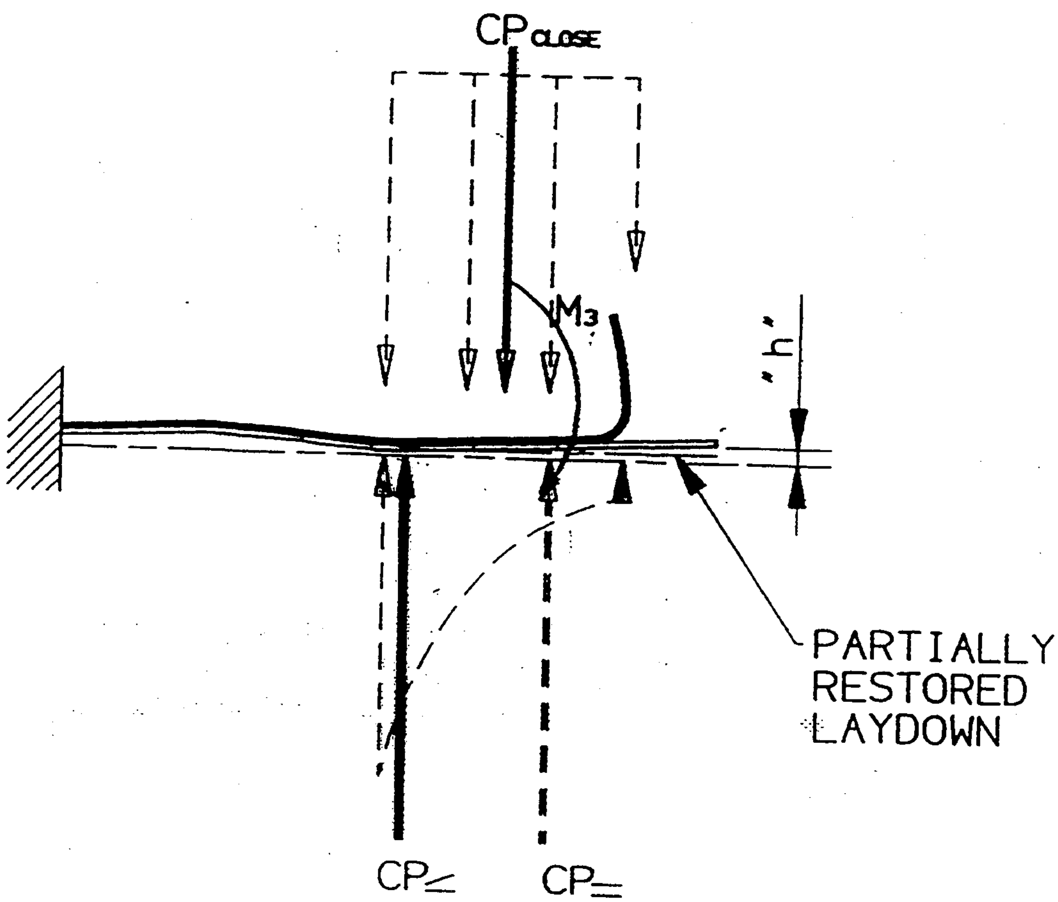
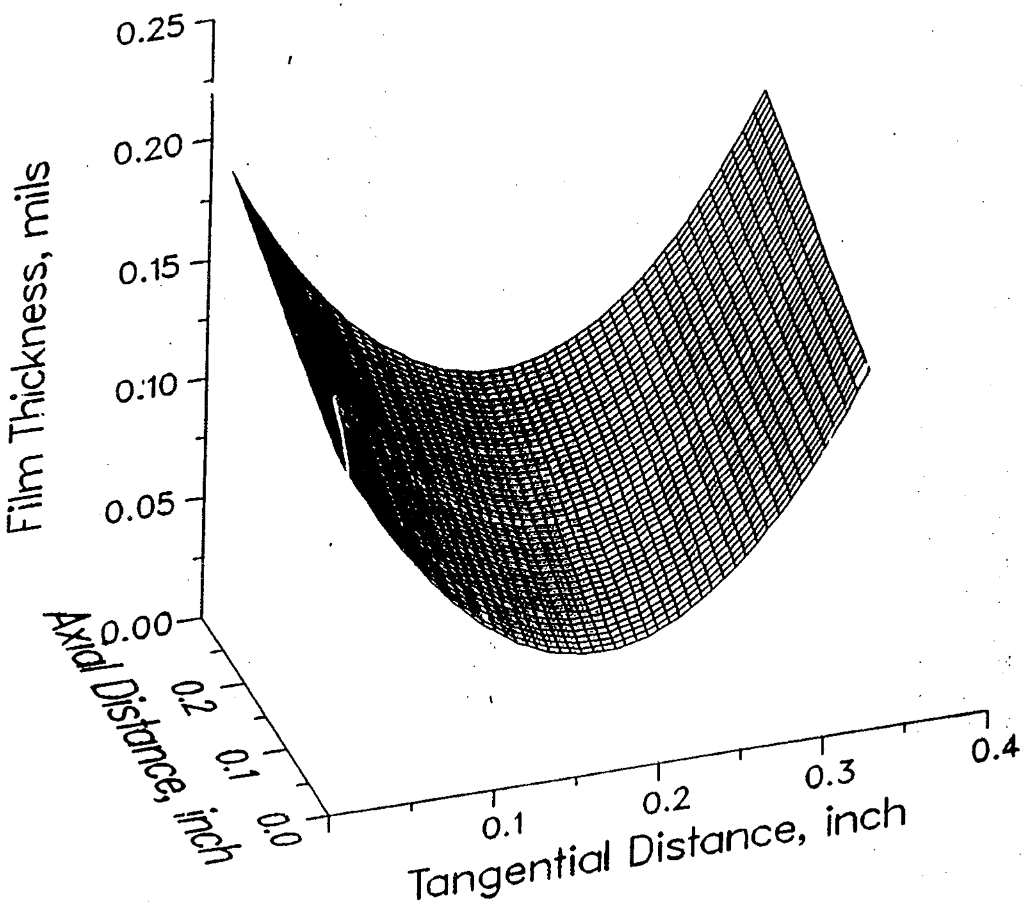
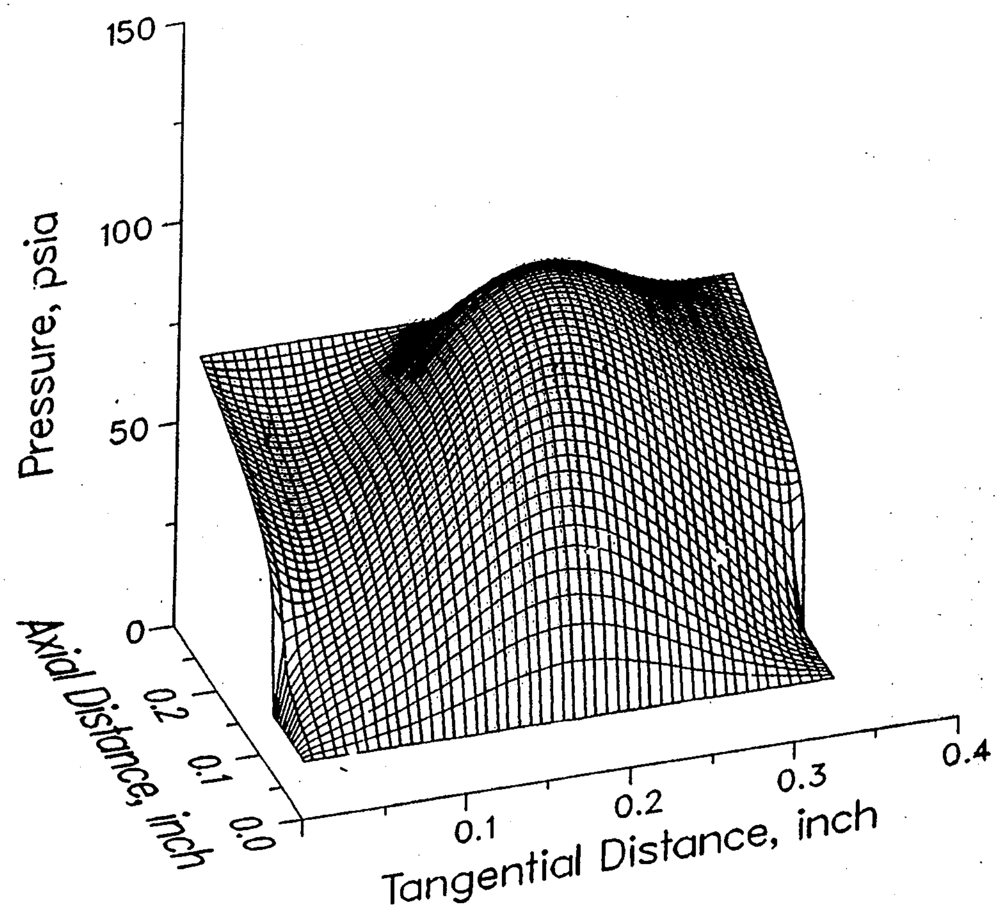


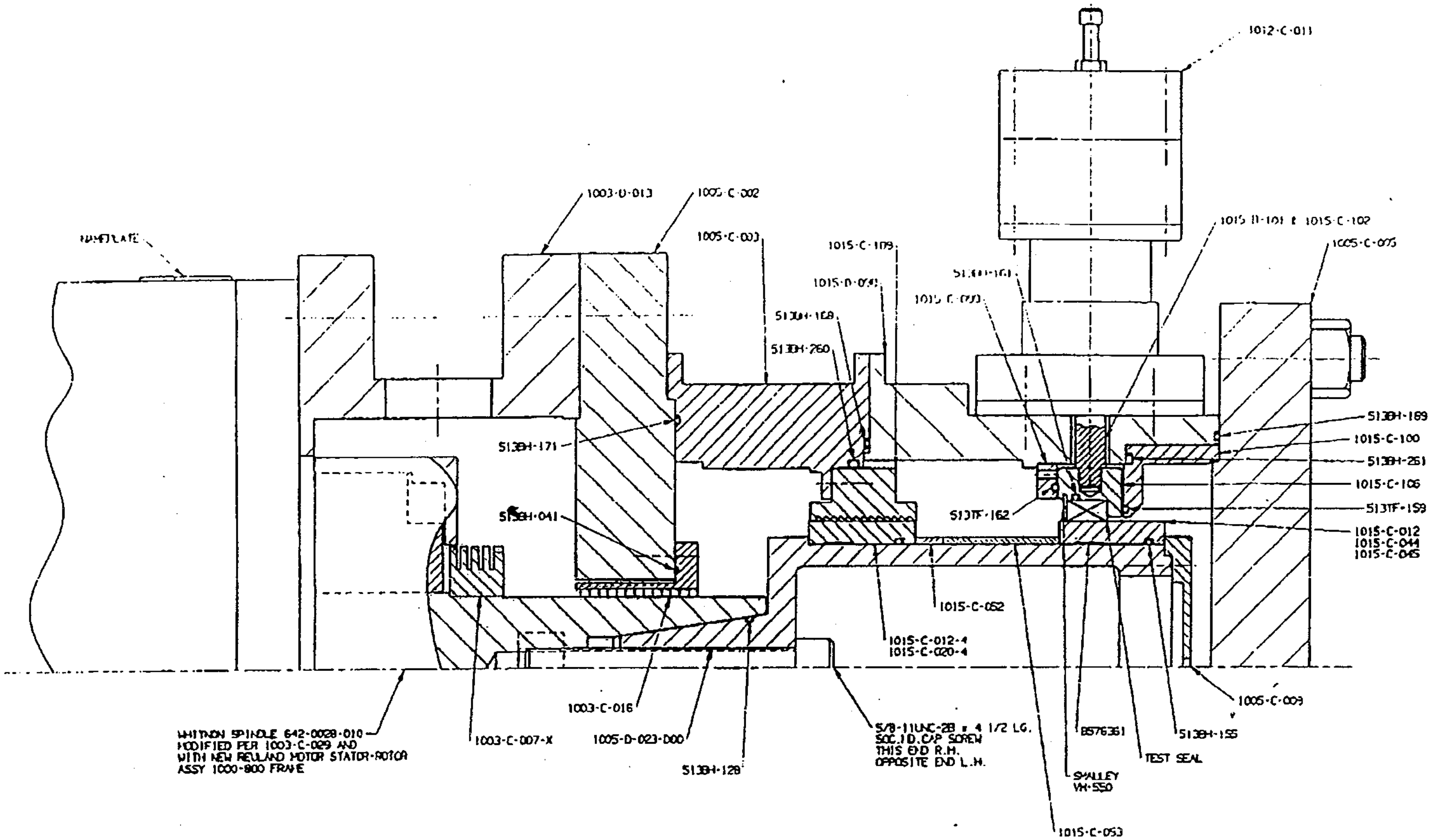
FIGURE 7: DIVERGING WEDGE CREATES MOMENT "M3", WHICH WILL CAUSE A PARTIAL RESTORATION OF LAYDOWN.

FILM THICKNESS DISTRIBUTION



PRESSURE DISTRIBUTION





UNLESS OTHERWISE SPECIFIED
ALL DIMENSIONS IN INCHES
REMOVE ALL BURRS
BREAK ALL CORNERS .005-.015
FILLET RADII .005-.015
DIM. TOL.: FRACT. +/- .015
.XX DEC. +/- .010 125/RMS
.XXX DEC. +/- .005 ALL OVER
ANGLES +/- 1/2°

MATERIAL

PROCESS

PART NAME: COMPLIANT SEAL SET-UP ASSY

PART NO.: 1015-D-275

DRAWN: D. MORIN

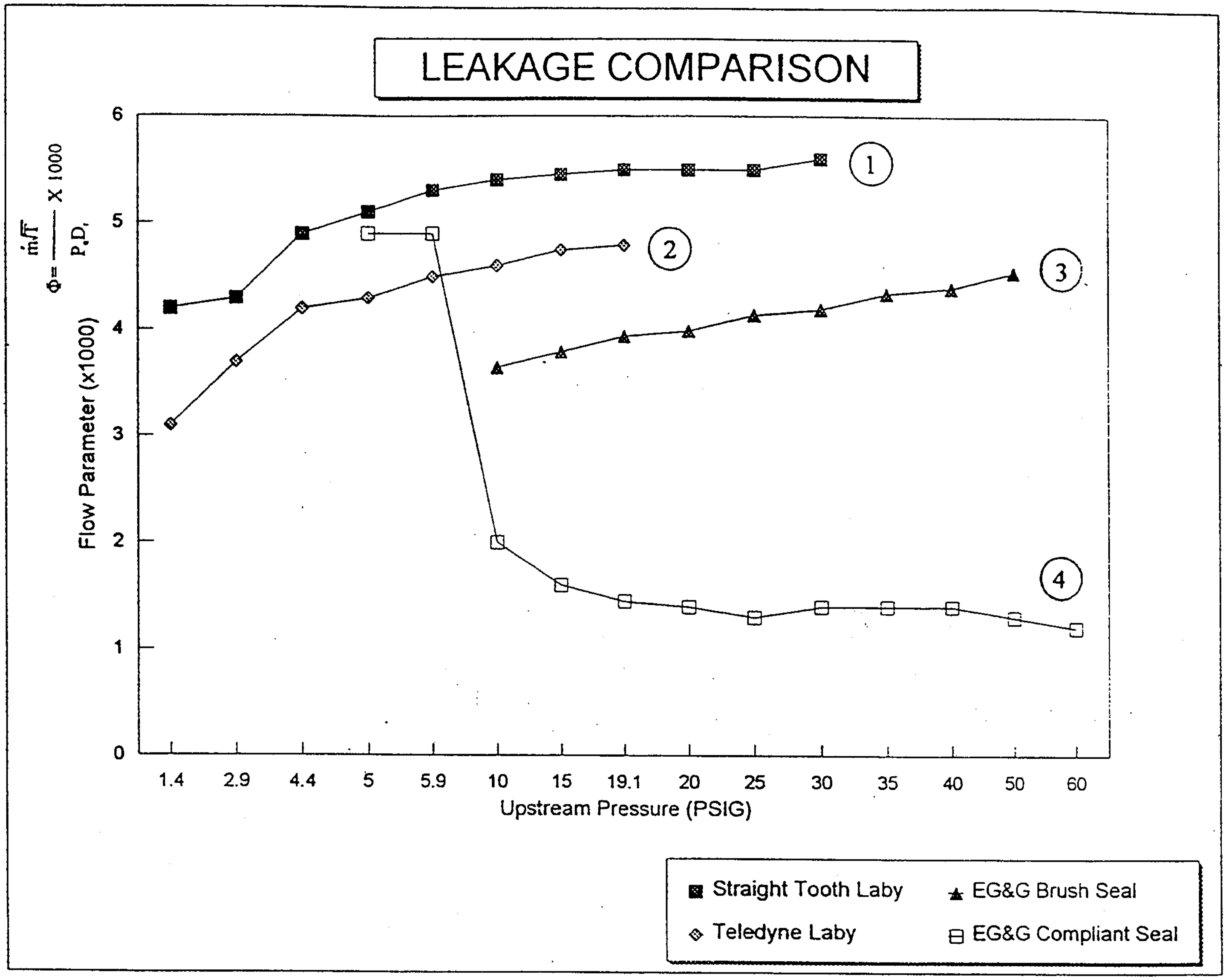
DATE: 6-12-95

SCALE: FULL

THIRD ANGLE PROJECTION

APPRVD.

EG&G MECHANICAL COMPONENTS GROUP
RESEARCH AND DEVELOPMENT

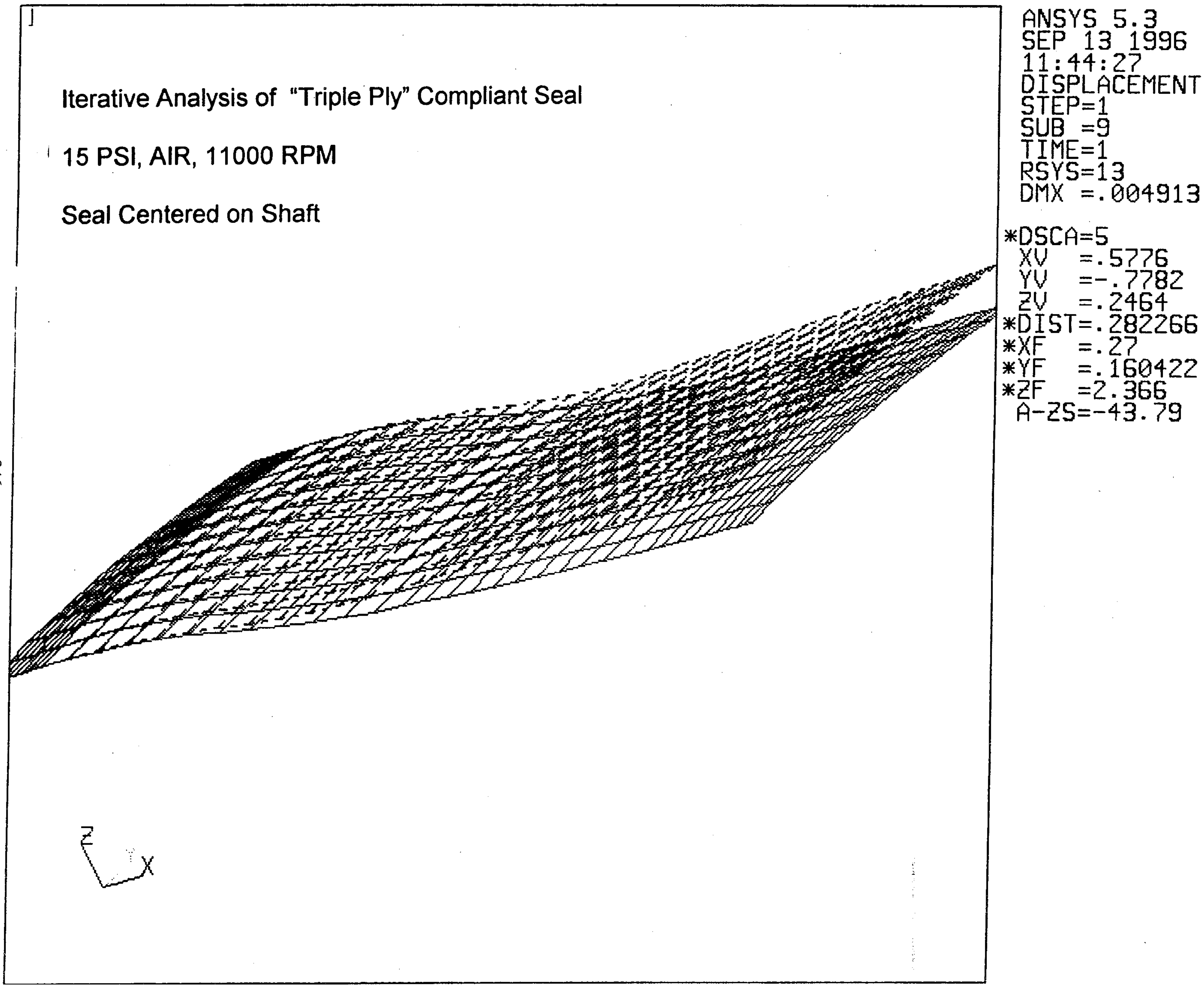


1. Straight Tooth Labyrinth Seal
   2.751 Diam. , .125 Pitch, 20 Teeth, .006" Radial Clearance

2. Labyrinth Seal, Teledyne Experimental Data
   30,000 RPM, 600°F, Estimated Operating Clearance - .006".

3. EG&G Experimental Brush Seal
   Surface Speed - 900 ft/sec, 420°F.

4. EG&G Experimental "Triple-ply" Seal
   45 Fingers, 4.760 Dia., 10,000 RPM


Iterative Analysis of "Triple Ply" Compliant Seal
15 PSI, AIR, 11000 RPM
Seal Centered on Shaft
ANSYS 5.3
SEP 13 1996
11:44:27
DISPLACEMENT
STEP=1
SUB =9
TIME=1
RSYS=13
DMX =.004913
*DSCA=5
XV =.5776
YV =-.7782
ZV =.2464
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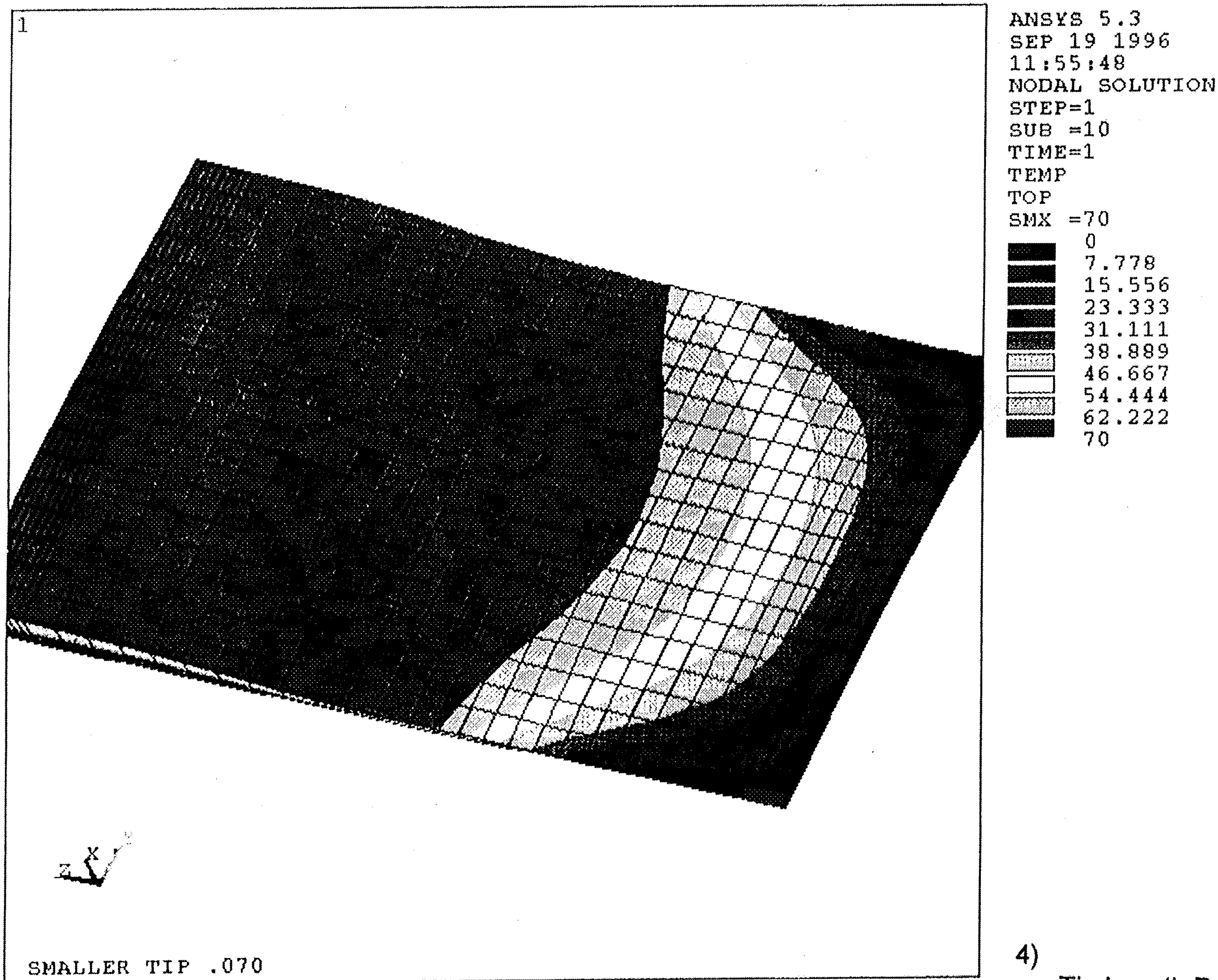
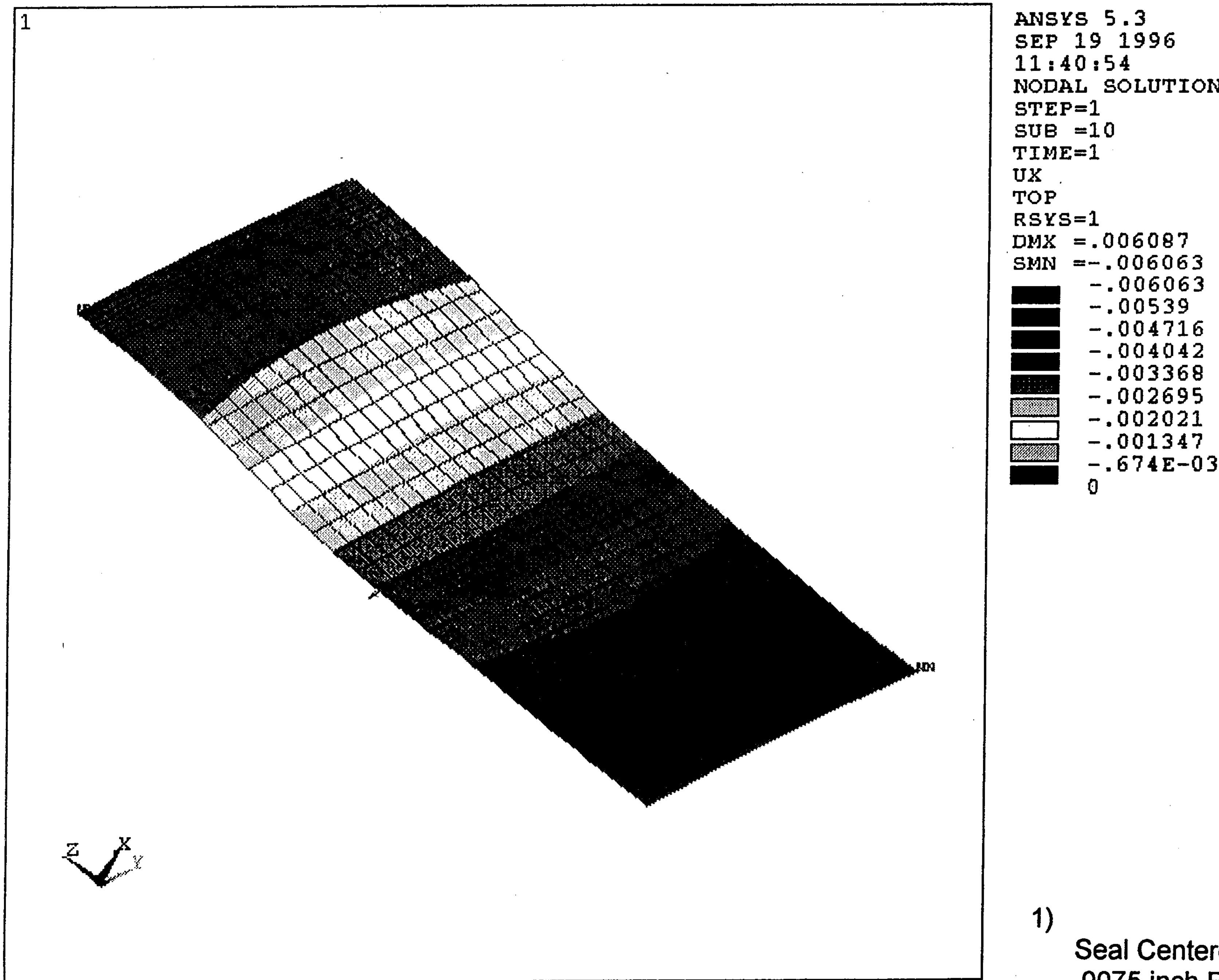
Z
Y
X



Iterative Analysis of <u>"Triple Ply" Compliant Seal</u>

70 PSIG, AIR, 20000 RPM

4) Tip Length Reduced
From .077 inch to .070 inch



1)

Seal Centered on Shaft
.0075 inch Radial Clearance

Iterative Analysis of <u>"Triple Ply" Compliant Seal</u>

70 PSIG, AIR, 20000 RPM